

Experimental investigation of the performance of an elbow-bend type heat exchanger with a water tube bank used as a heater or cooler in alpha-type Stirling machines

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ABSTRACT

In this work the effect of the elbow-bend geometry and the effect of the tube arrangement on the performance of air-to-water heat exchanger is studied experimentally. In elbow-bend heat exchanger, the direction of the working fluid is bended at 90 degrees to its inlet direction. The heating or cooling fluid flows inside straight tubes while the working fluid flows past the tubes along an elbow pass. Three different types of the geometry of the elbow with three different tube bank arrangements were studied. The results were plotted and analyzed to clarify the effects of the elbow-bend geometry, the tube bank arrangements and the dead volume in the heat exchanger on the heat transfer and pressure drop. Two empirical correlations were deduced for each design, one to predict the relation between Nusselt and Reynolds numbers, while the other relation is between the friction factor and Reynolds number. This work was done to select the more suitable design to be used as a heater or a cooler in Stirling machines.

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1. Introduction

Stirling engines can be operated using various heat sources. Referring to the literature of Stirling engines, different designs of heat exchangers are found. Some of them are explained below. A small gamma-type Stirling engine was tested by Iwamoto et al. [1]. In this engine the heat exchangers are located around the displacer cylinder. The heater and the cooler are of the fine-tube type. Hot water flows in the heater tubes. Coolant water flows in the cooler tubes. The working gas flows around these tubes. Hoshino [2] used twelve electrical heaters attached to a gamma-type Stirling engine heater head. The working gas is heated through annular flow passages of a groove-type heater; each of them has 25 plate fins around a cylindrical heater. The cooler is of the shell-and-tube type, which has 322 tubes of 1 mm inner diameter. A heat input system which is composed of a heater and a burner as a newly devised heat exchanger in a Beta-type Stirling engine was designed and tested by Chang [3]. The heater likes a U-cup and it has slits which play the role of the heater tubes on the outer wall of the cup. A 200 W domestic free-piston Stirling electric power

system with multi-fuel capability was designed by Biao et al. [4]. The engine was fitted with a fine heater and an annular shaped clearance regenerator. The cylinder head was provided with vertical fins. At operation, the hot gases flow along the vertical fins, and the heat was supplied to the helium working gas through these fins. A thermodynamic analysis of a Stirling engine including dead volumes of hot space, cold space and regenerator was carried out by Kongtragool and Wongwises [5]. The analysis indicated that the engine net work is affected by only the dead volume while the efficiency is affected by both the regenerator effectiveness and dead volume. The heat transfer characteristics and pressure drop of block-type heat exchangers were experimentally investigated by El-Ehwany et al. [6]. The experimental data were compared to get the most suitable block-type heat exchanger to be used as a cooler and a heater in Stirling machines. Generalized correlations of heat transfer and friction factor for air-side plain fin-and-tube heat exchangers were reported by Wang et al. [7], where 31 samples of fin-and-tube heat exchangers were used to develop the correlations. An experimental study to investigate the heat transfer and friction characteristics of fin-and-tube heat exchangers was carried out by Yan and Sheen. [8]. The results were presented as plots of friction factor and Colburn factor versus Reynolds number. Kim et al. [9] used a multiple regression technique to correlate the data of 47 sets of plain fin-and-tube heat

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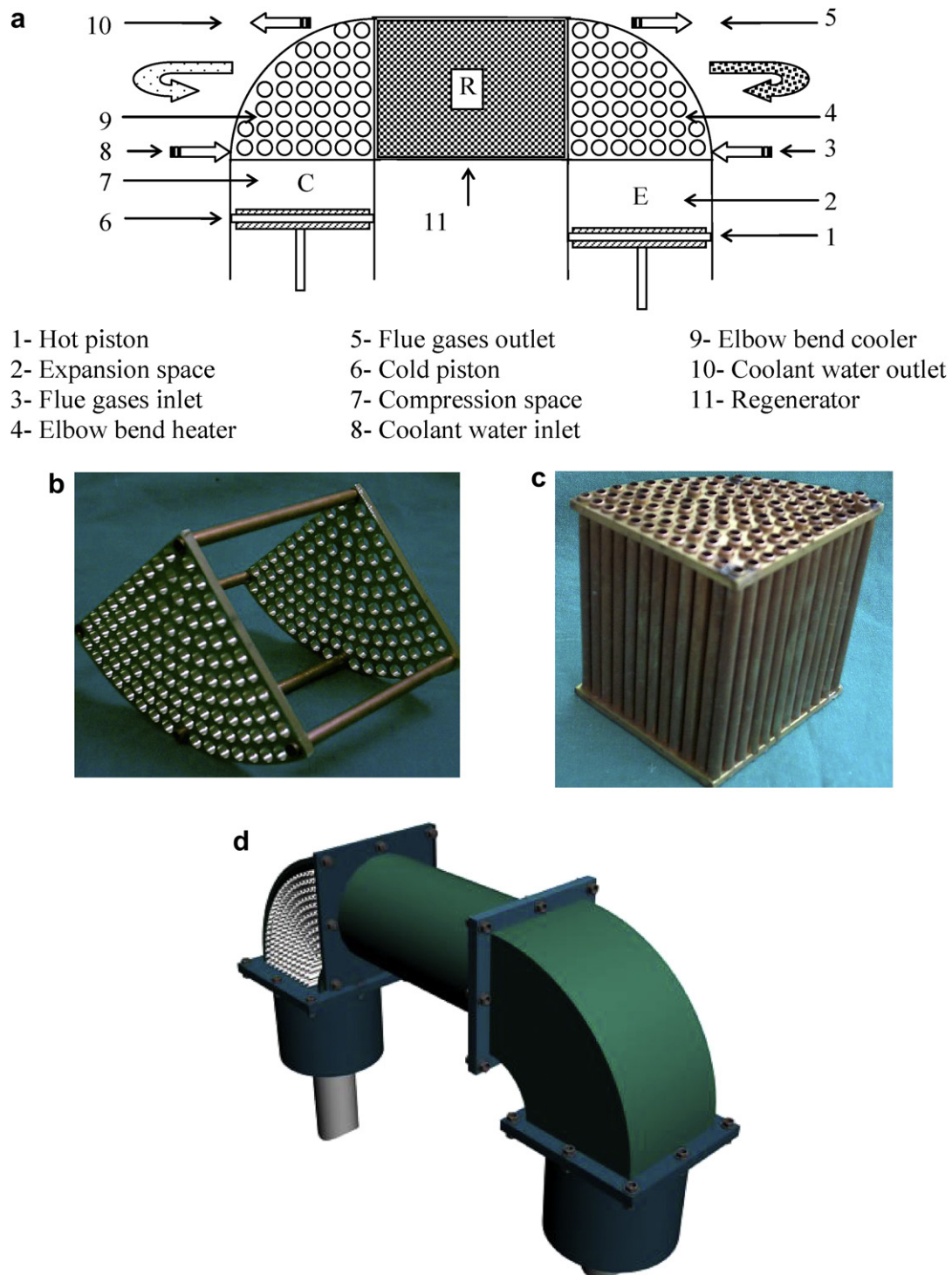


Fig. 1. a. Schematic diagram of the main parts of an alpha-type Stirling engine. b. Elbow-bend heat exchanger assembly. c. Elbow-bend heat exchanger. d. Isometric of the main parts of an alpha-type Stirling engine with elbow-bend heat exchangers.

exchangers with staggered tube arrangements to develop the heat transfer and friction correlations. The geometry and the dimensions of the heat exchangers have a strong influence on the performance of Stirling engines. Therefore, the processes in the heat exchangers should be carefully considered in the modeling of the engine processes. Modern second-order thermodynamic model was used to simulate Stirling cycle to calculate the heat transfer and pressure drop during the engine cycle, and as

a consequence, the obtained information is in turn used to rectify the prediction of the engine performance as provided by Mahkamov and Ingham [10]. Moreover, recently, 2 and 3-dimensional CFD models have been developed for the modeling of the heat and mass transfer processes in heat exchangers of Stirling engines, which were considered as integral parts of the gas circuit as provided by Mahkamov [11,12]. The heat exchangers were also analyzed separately as given by Mahkamov and Ingham [13].

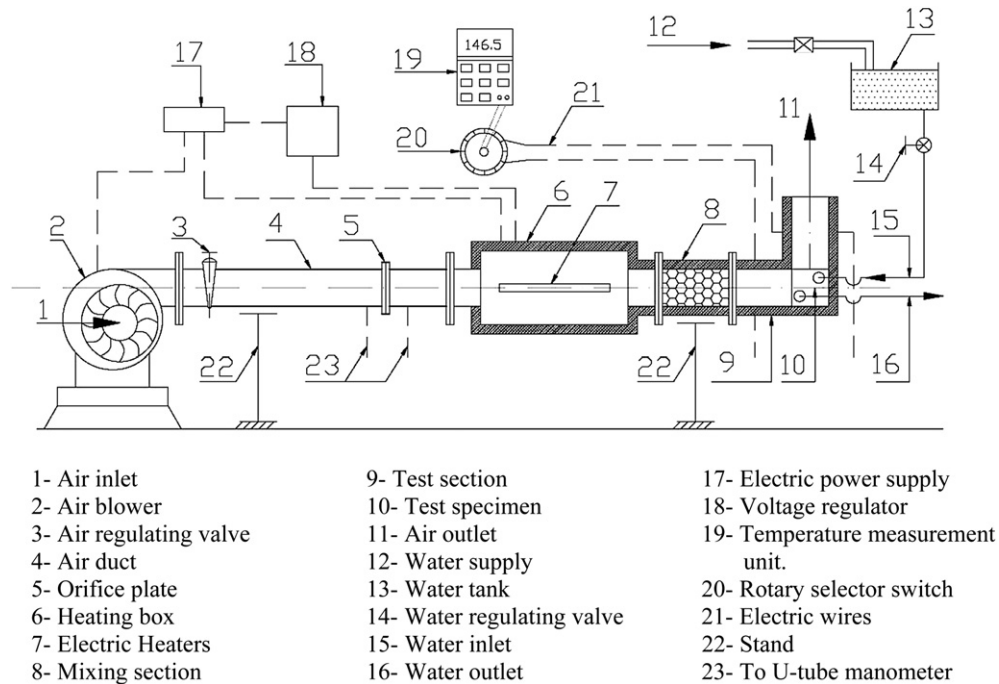


Fig. 2. Experimental test rig.

The conventional designs of the alpha-type Stirling engine with twin parallel cylinders may have plain heater and cooler or tubular heater and cooler. For plain heat exchanger, insufficient heat transfer rates were observed and for tubular heat exchangers, the tubes are bent as a quadrant of a circle and the pressure drop is increased [14]. The suggested design aims at using straight tubes instead of bent ones. This restrains the flowing of the heat carrier fluid and the cooling agent through the tubes and the working fluid flows past the tubes as shown in Fig. 1. The major disadvantage of the present suggestion is that the dead volume of the engine could be increased compared with the traditional ones. However, the decrease in pressure drop is an important factor in the present aspect.

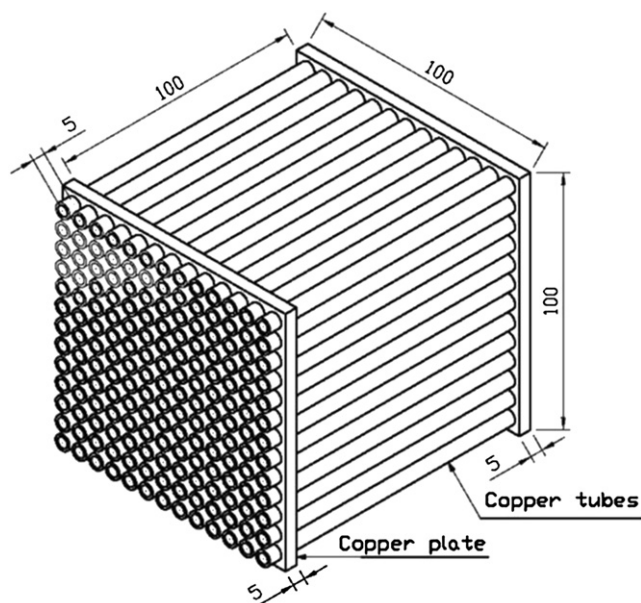


Fig. 3. Isometric of the test specimen No. II.

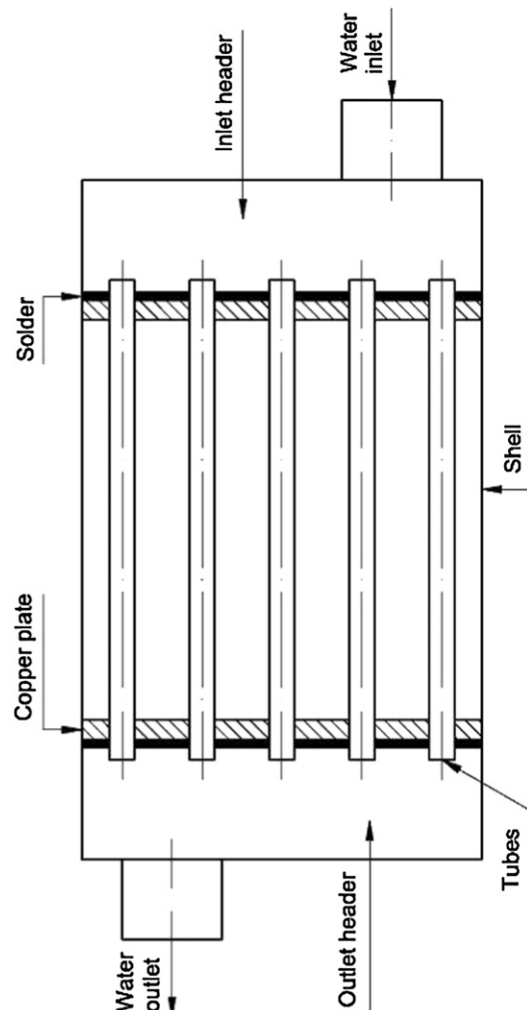


Fig. 4. Cross section of the test specimen inside the shell.

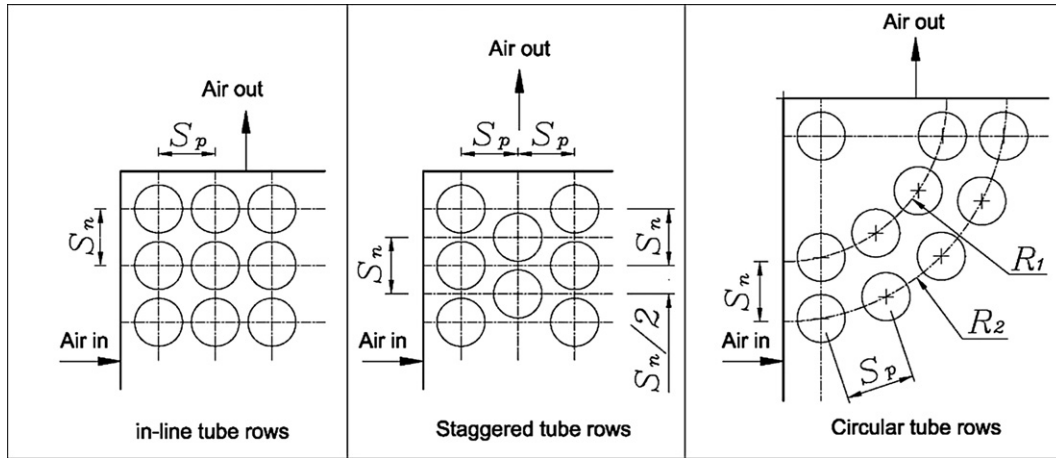
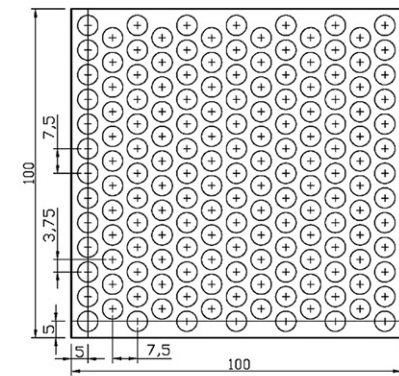


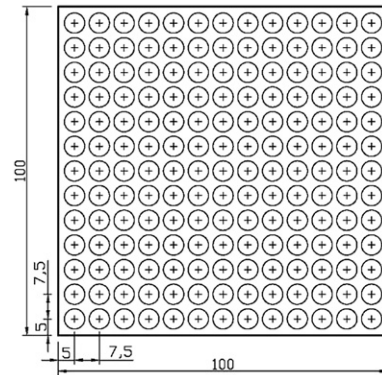
Fig. 5. Schematic diagram of the tube arrangements and its center distances.

In the present work, an elbow-bend shell-and-tube heat exchanger was introduced as a new design for the heater and cooler of alpha-type Stirling engines. The elbow-bend heat exchanger consists of a shell in the form of a right-angle elbow

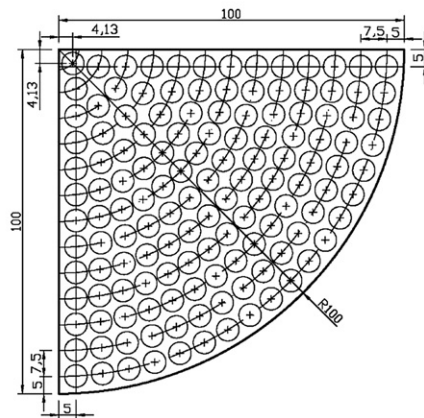
and a tube bank fitted perpendicular to the flow in the elbow. Eight specimens of this heat exchanger were designed, manufactured and tested experimentally. The experimental investigation is revealed of the influence of the elbow-bend geometry,



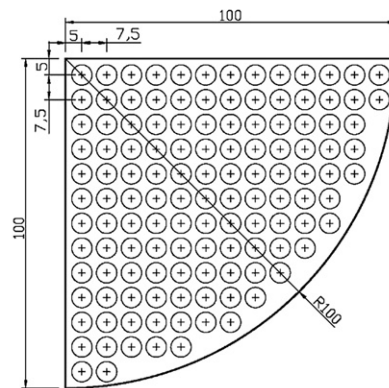
Specimen No. (I), 163 tubes (staggered arrangement), $\psi = 48.4\%$, $S_v = 325.17 \text{ m}^{-1}$
 $S_n = S_p = 7.5 \text{ mm}$



Specimen No. (II), 169 tubes (in-line arrangement), $\psi = 46.5\%$, $S_v = 337.14 \text{ m}^{-1}$
 $S_n = S_p = 7.5 \text{ mm}$

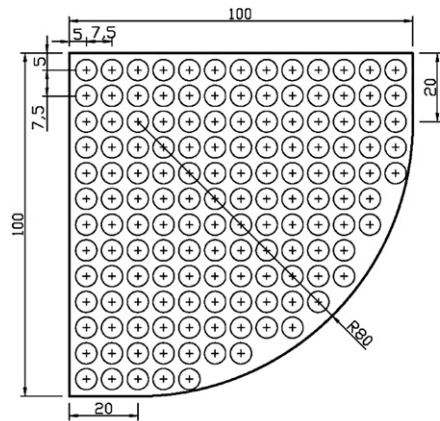


Specimen No. (III), 136 tubes (circular arrangement), $\psi = 45.2\%$, $S_v = 345.44 \text{ m}^{-1}$
 19 tubes in the outer row,
 $S_n = 7.5 \text{ mm}$, $S_p = 7.62 \text{ mm}$

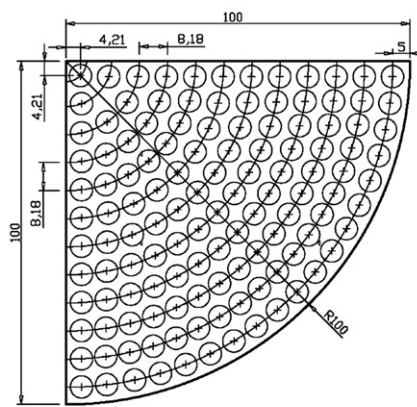


Specimen No. (IV), 125 tubes (in-line arrangement), $\psi = 49.6\%$, $S_v = 317.5 \text{ m}^{-1}$
 $S_n = S_p = 7.5 \text{ mm}$

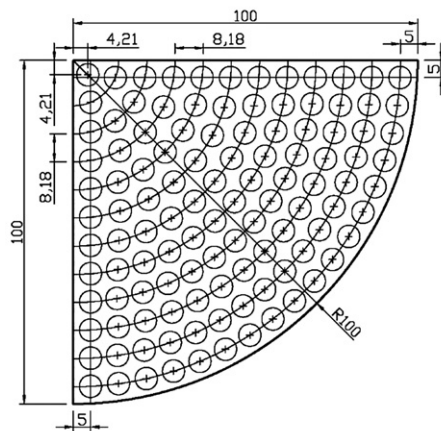
Fig. 6. Geometrical specifications of the test specimens No. I, II, III & IV.



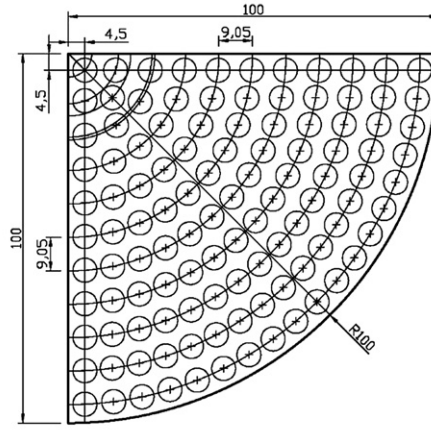
Specimen No. (V), 142 tubes (in-line arrangement), $\psi = 47.9\%$, $S_v = 328.38 \text{ m}^{-1}$
Header: $2 \times 20 \times 100$ + quadrant of 80 mm radius, $S_n = S_p = 7.5 \text{ mm}$



Specimen No. (VI), 124 tubes (circular arrangement), $\psi = 50.0\%$, $S_v = 314.96 \text{ m}^{-1}$
19 tubes in the outer row, $S_n = 8.18 \text{ mm}$, $S_p = 7.78 \text{ mm}$



Specimen No. (VII), 113 tubes (circular arrangement), $\psi = 54.4\%$, $S_v = 287.02 \text{ m}^{-1}$
18 tubes in the outer row, $S_n = 8.18 \text{ mm}$, $S_p = 8.51 \text{ mm}$



Specimen No. (VIII), 111 tubes (circular arrangement), $\psi = 55.2\%$, $S_v = 281.94 \text{ m}^{-1}$
19 tubes in the outer row, $S_n = 9.05 \text{ mm}$, $S_p = 7.78 \text{ mm}$

Fig. 7. Geometrical specifications of the test specimens No. V, VI, VII & VIII.

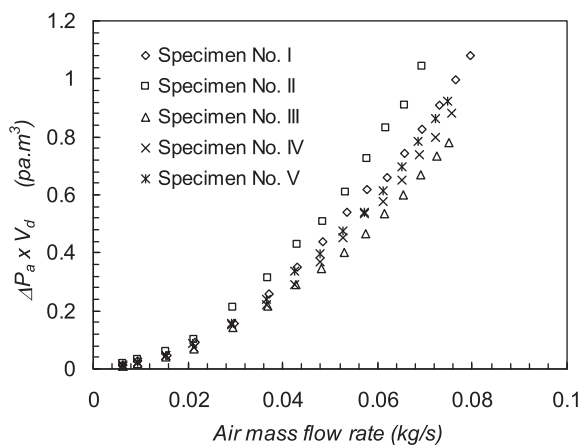


Fig. 8. Variation of the quantity $(\Delta p_a \times V_d)$ versus air mass flow rate.

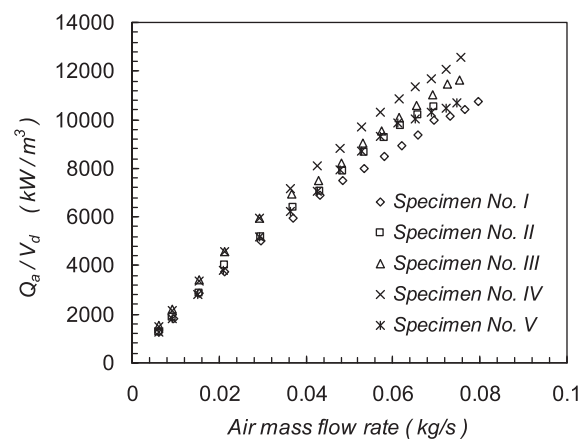


Fig. 9. Variation of the quantity (Q_a/V_d) versus air mass flow rate.

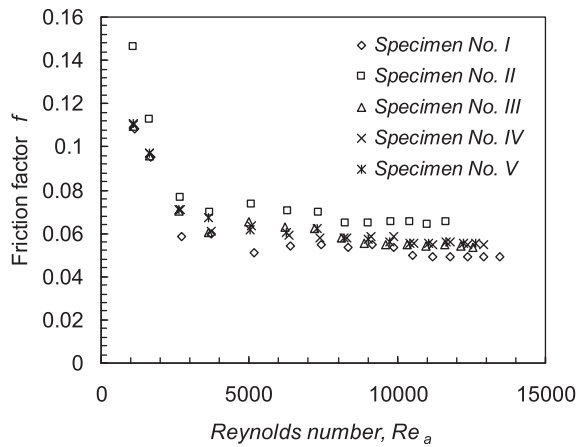


Fig. 10. Variation of the friction factor versus Reynolds number.

tube bank arrangement, and dead volume (difference between the elbow volume and the volume occupied by the tubes) on the thermal performance and pressure drop of the elbow-bend heat exchanger.

2. Experimental apparatus

Fig. 2 shows a schematic diagram of the test rig used in this work. It consists of a wind tunnel system forcing hot air past the outside surface of the tube bank by a 3 HP centrifugal fan. The air duct was a circular duct of 52.3 mm or 104.6 mm inner diameter. The air flow rate was measured by using one of the two orifice plates (one for each duct) that were designed and manufactured according to BSI data catalog [15]. The air flow rate was controlled by a regulating gate valve. The air was heated by five heaters having a total power of 10.75 kW in a heating box. Four heaters were connected directly to the main supply, while the last one was connected through a voltage regulator. The air temperature was adjusted at 146 ± 1.5 °C at the inlet of the test section. The water was supplied through a tank with a float supported at 7 m above the test section to sustain a constant water flow rate during the experiments. The water flow rate was measured using a tank having a definite volume and a stop watch. The temperature was measured by thirty-one pre-calibrated T-type thermocouples. The inlet and exit temperatures of air were measured by two

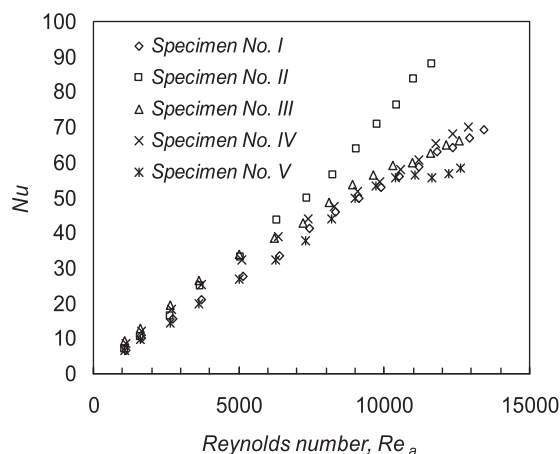


Fig. 11. Variation of Nusselt number versus Reynolds number.

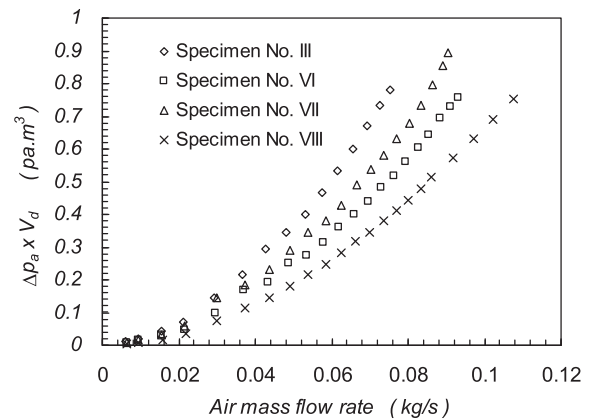


Fig. 12. Variation of the quantity ($\Delta p_a \times V_d$) versus air mass flow rate.

thermocouple meshes. Each mesh consists of 13 thermocouples connected in parallel to evaluate the average temperature [16]. The inlet and exit temperatures of the cooling water were measured by two thermocouples; one at inlet and the other at the water exit. The tube surface temperature was measured by using two thermocouples. The air temperature at the orifice plate was measured by one thermocouple. The thermocouples were connected to a digital temperature recorder model 922 (Manufacturer: Maxtec Int. Co.). It has 0.1 °C resolution in a range up to 200 °C.

3. Test specimens

Referring to Fig. 1b, twin quadrants of a circular disc having 5 mm thickness, were drilled using the plain radial drilling machine and the distribution of holes was accurately carried out by using the universal dividing head. A certain number of copper tubes having 6.35 mm outer diameter, 0.4 mm wall thickness and 120 mm length were assembled to the twin quadrant by welding while keeping 100 mm distance apart between the quadrants as shown in Fig. 1c. Referring to Fig. 3, an isometric view of the specimen No. II is clear. Referring to Fig. 4, a square or a quadrant circle header was assembled to the twin discs. The tube arrangement allows the variation of both normal and parallel pitches s_n and s_p as shown in Fig. 5. In the vision of previous description, eight test specimens were designed and manufactured. The complete geometrical specifications of each one is given in Figs. 6 and 7.

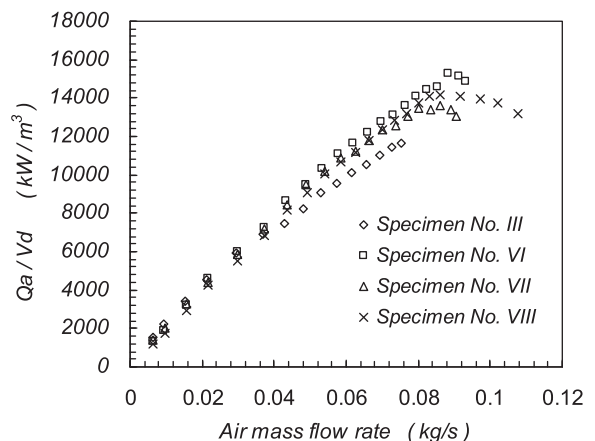


Fig. 13. Variation of the quantity (Q_a / V_d) versus air mass flow rate.

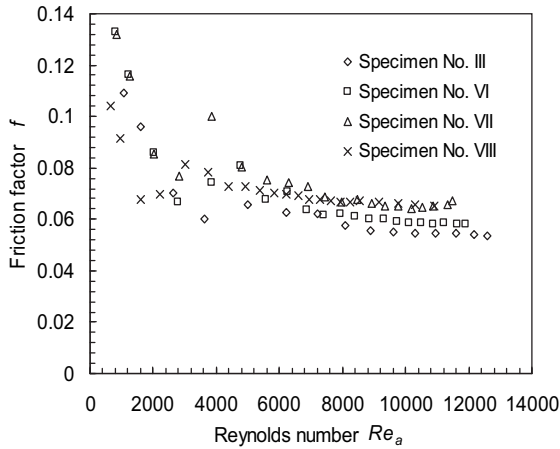


Fig. 14. Variation of the friction factor versus Reynolds number.

4. Experimental data reduction

The analysis of the present work is concerned with the calculation of non-dimensional numbers for each experiment and deducing generalized correlations to explain the case study. The following equations were used in the calculations:

$$h_a = Q_a / [A_a (T_{a,avg} - T_{s,avg})] \quad (1)$$

$$h_w = Q_w / [A_w (T_{s,avg} - T_{w,avg})] \quad (2)$$

$$Nu_a = h_a d_o / k_a \quad (3)$$

$$Nu_w = h_w d_i / k_w \quad (4)$$

$$G_{max} = \dot{m}_a / A_{min} \quad (5)$$

$$Re_a = \dot{m}_a d_o / [A_{min} \mu_a] \quad (6)$$

$$A_{min} = L(B - N_n d_o) \quad (7)$$

$$f = \Delta p_a / \rho_a \text{avg} / [2 G_{max}^2 N_p] \quad (8)$$

$$\varepsilon = Q_{avg} / Q_{max} = Q_{avg} / [(m \cdot C_p)_a (T_{i,a} - T_{i,w})] \quad (9)$$

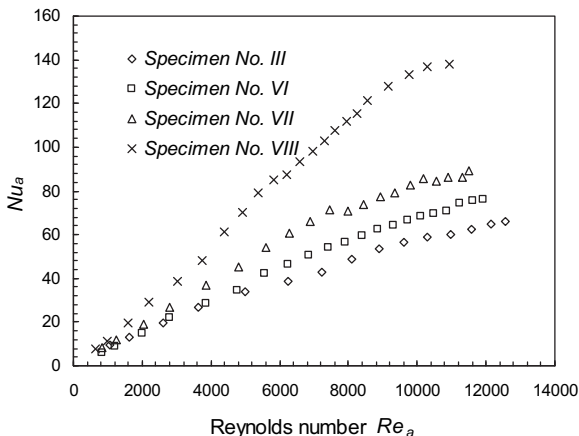


Fig. 15. Variation of Nusselt number versus Reynolds number.

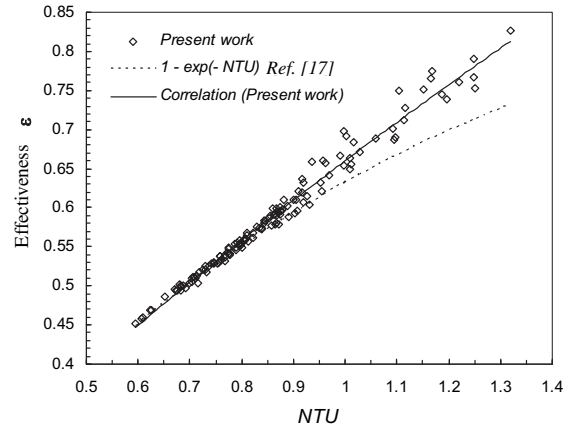


Fig. 16. Comparison of present and previous studies. ($NTU - \varepsilon$).

$$NTU = U A / C_{min} \quad (10)$$

$$C^* = C_{min} / C_{max} = (\dot{m} \cdot C_p)_a / (\dot{m} \cdot C_p)_w \quad (11)$$

$$\psi = V_d / V_t \quad (12)$$

$$S_v = A_a / V_t \quad (13)$$

The properties of the air and the water were found at the film temperature [17].

5. Results and discussion

In Stirling machines, increasing both or one of the dead volume and pressure drop of the heater or cooler causes deteriorations in the power and efficiency [7]. In the heat exchangers it is usually expected that the pressure drop increases as the dead volume decreases and visa versa. So, in the present work it is convenient to select the specimens that are having low ($\Delta p_a \times V_d$) (pressure drop times dead volume) and high (Q_a / V_d) (heat transfer rate divided by dead volume). The quantities ($\Delta p_a \times V_d$) and (Q_a / V_d) were compared for the specimens No. I, II, III, IV and V (the five specimens which have equal normal and parallel pitches) versus the air flow rate in Figs. 8 and 9. It is clear that the quantity ($\Delta p_a \times V_d$) tends to increase with the increase of the air mass flow rate for the five specimens. The specimen No. III exhibits the lowest value of ($\Delta p_a \times V_d$). The reduction in ($\Delta p_a \times V_d$) is due to its small dead volume V_d , and it has a circular air pass about the tubes which minimizes eddies formation and as a consequence the pressure drop reduces. The specimen No. IV exhibits the highest value of (Q_a / V_d). This is due to the streamlines of air not straight due to the circular pass of air through an in-line arrangement of tubes which provides different angles of attack between the air and the tubes. Referring to Fig. 10, the friction factor f is compared for the specimens No. I, II, III, IV and V versus Reynolds number. The specimen No. I exhibits the lowest friction factor f , due to minimum (Δp_a) for the same air flow rates. Fig. 11 implies that the specimen No. I gives higher Nusselt number compared with the other ones.

From the pervious discussion, it is better to select the specimen No. III to be compared with the specimens No. VI, VII, and VIII. The quantities ($\Delta p_a \times V_d$) and (Q_a / V_d) are compared for the specimens No. III, VI, VII, and VIII versus the air flow rate in Figs. 12 and 13. It is clear that the specimen No. VIII has the lowest ($\Delta p_a \times V_d$). Also, the specimens No. VI, VII, and VIII have approximately the same values of (Q_a / V_d), that are slightly greater than that of specimen No. III.

Table 1
Constants C_1 , C_2 , C_3 , A and B.

Specimen No.	Range	C_1	C_2	C_3	A	B
I	$1100 \leq Re_a < 2700$	0	-3.174×10^{-5}	0.1453	9.327×10^{-3}	0.9389
	$2700 \leq Re_a < 10500$	6.476×10^{-11}	-1.646×10^{-6}	0.06239		
	$10500 \leq Re_a < 10500$	0	-2.110×10^{-7}	0.05189		
II	$1100 \leq Re_a < 2700$	0	-4.313×10^{-5}	0.1892	3.691×10^{-3}	1.072
	$2700 \leq Re_a < 8200$	-1.480×10^{-10}	9.567×10^{-8}	0.07543		
	$8200 \leq Re_a < 11700$	0	1.746×10^{-8}	0.06486		
III	$1090 \leq Re_a < 2600$	0	-2.517×10^{-5}	0.1364	3.598×10^{-2}	0.7995
	$2600 \leq Re_a \leq 8900$	-1.379×10^{-10}	1.297×10^{-7}	0.06832		
	$8900 \leq Re_a \leq 12600$	0	-3.680×10^{-7}	0.05856		
IV	$1100 \leq Re_a \leq 2700$	0	-2.474×10^{-5}	0.1376	2.203×10^{-2}	0.8520
	$2700 \leq Re_a \leq 10600$	2.387×10^{-10}	-4.601×10^{-6}	0.07916		
	$10600 \leq Re_a \leq 12900$	0	-6.525×10^{-8}	0.05590		
V	$1100 \leq Re_a \leq 2600$	0	-2.525×10^{-5}	0.1377	1.149×10^{-2}	0.9112
	$2600 \leq Re_a \leq 9700$	1.881×10^{-10}	-4.238×10^{-6}	0.08029		
	$9700 \leq Re_a \leq 12600$	0	-3.207×10^{-8}	0.05596		
VI	$800 \leq Re_a \leq 2800$	0	-3.375×10^{-5}	0.1580	1.116×10^{-2}	0.9478
	$2800 \leq Re_a \leq 9700$	-3.795×10^{-10}	2.546×10^{-6}	0.06765		
	$9700 \leq Re_a \leq 11900$	0	-2.348×10^{-7}	0.06107		
VII	$800 \leq Re_a \leq 2800$	0	-2.875×10^{-5}	0.1552	1.776×10^{-2}	0.9202
	$2800 \leq Re_a \leq 8000$	-1.018×10^{-9}	7.084×10^{-6}	0.07237		
	$8000 \leq Re_a \leq 11500$	0	-3.628×10^{-7}	0.06937		
VIII	$600 \leq Re_a \leq 2200$	0	-3.872×10^{-5}	0.1289	7.502×10^{-3}	1.069
	$2200 \leq Re_a \leq 7900$	-6.458×10^{-10}	5.183×10^{-6}	0.06403		
	$8000 \leq Re_a \leq 10,900$	0	-5.862×10^{-7}	0.07171		

Referring to Fig. 14, the specimen No. III has the lowest friction factor f . The specimen No. VIII has the highest Nusselt number compared with the other specimens at the same Reynolds number and it shows adequate accordance with the diagram of the heat transfer as shown in Fig. 15. The effectiveness (ε) was determined from equation (9) in which the heat transfer rate Q_{avg} was determined as the average of the air and water side values. Referring to Fig. 16, the effectiveness (ε) was plotted versus the number of transfer units (NTU). This was done using the experimental results of all the heat exchangers which were tested in the present work. The relation was correlated as follows:

$$\varepsilon = 0.6602 \times NTU^{0.7527} \quad (14)$$

The friction factor can be correlated as:

$$f = C_1 \times Re_a^2 + C_2 \times Re_a + C_3 \quad (15)$$

Nusselt Number can be correlated as:

$$Nu = A \times Re_a^B \quad (16)$$

The constants C_1 , C_2 , C_3 , A and B are shown in Table 1.

6. Comparison of present and previous studies

Fig. 16 shows a comparison between the values of the effectiveness (ε) for all the heat exchangers in the present work and those calculated from the correlation $\varepsilon = 1 - e^{-NTU}$ in Ref. [17] assuming that $C^* \approx 0$. It can be seen that the experimental results

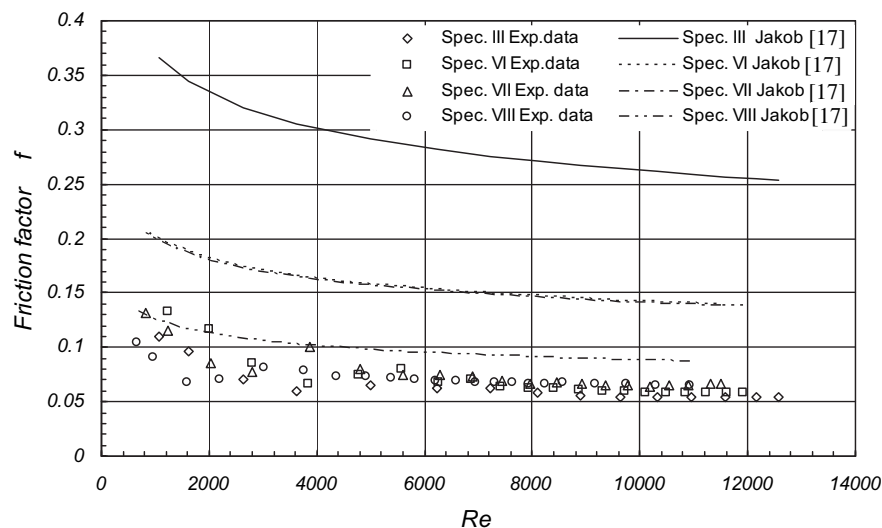


Fig. 17. Comparison of present and Jakob's works. ($Re - f$).

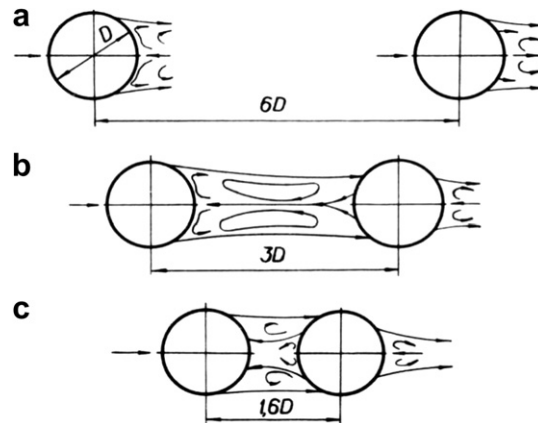


Fig. 18. Flow pattern past an in-line-tube bank [18].

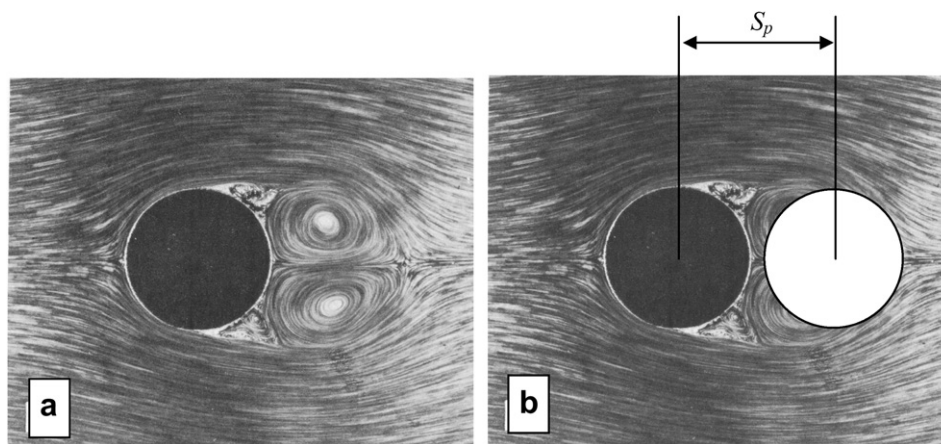


Fig. 19. a. Flow pattern past one tube [14]. b. Location of the next tube (white) in the present work.

are very close to that in the previous work for $NTU \leq 0.75$. For higher values of NTU , the experimental results show higher values of effectiveness.

Fig. 17 shows a comparison between the empirical correlation of the friction factor f which is given in Ref. [17] for in-line arrangements and the friction factor f of the specimens No. III, VI, VII, VIII. It is clear that the results of Ref. [17] have higher f than

the present data. For an in-line tube bank, the location of the impact point depends on s_p/d and Reynolds number. For $s_p/d = 6$ the point of attack and the point of impact coincide at $\phi = 0.0^\circ$, while for $s_p/d = 6$, the point of impact is shifted to $\phi = 75^\circ$, as shown in Fig. 18 as provided by Kakac et al. [18].

The flow pattern past a tube is indicated by the image presented in Fig. 19a by Fraas and Ozisik [19]. Referring to this image, it is clear that, the eddy formation extended to a distance equals the tube diameter and hence the next tube should be located at a distance of $S_p < 1.6d$ as shown in Fig. 19b. In Fig. 19b, the eddy formation will be dissipated and the point of impact is shifted to $\phi = 90^\circ$ and as a consequence, there are no eddies on the rear surface of the first tube and no impact force on the front surface of the next one. The same can be applied on any two successive tubes. Then the intermediate surfaces of the tubes have no effect. The surfaces of the tubes in the longitudinal row are expected to be as a corrugated sheet as shown in Fig. 20. As a result, the pressure drop will be decreased. This is due to the values of s_p/d are ranging from 1.20 to 1.34 in the present work. This explains why the friction factor in the present work is less than that calculated from the correlation of Jakob, [17].

7. Conclusions

The following conclusions were made from the experimental tests of the air-side elbow-bend geometry and water tube bank arrangements.

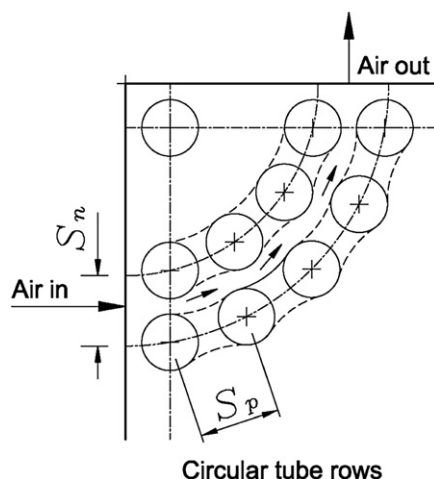


Fig. 20. Expected imaginary corrugated sheet of the tubes.

1. The circular arrangement of the tubes in elbow-bend heat exchangers is better than both in-line and staggered arrangements if both arrangements are having the same number of tubes.
2. The number of tubes does not have significant effect on the pressure drop s_p/d is less than 1.34.
3. The tube bank arrangement through elbow-bend has a significant effect on the pressure drop. It increases if s_n decreases and it also increases if s_p increases.
4. General correlations were deduced for the studied geometries of such heat exchangers that may be used in the design of alpha-type Stirling machines.

Nomenclature

A	surface area, m^2
B	heat exchanger width, m
C	heat capacity rate, J/K
C_p	specific heat at constant pressure, J/kg K
d	tube diameter, m
f	friction factor
G	mass flux of the air based on the minimum flow area, kg/s
h	heat transfer coefficient, $W/m^2 K$
j	Colburn factor
k	thermal conductivity, W/m K
L	effective length of tube, m
$\dot{m}$	mass flow rate, kg/s
N	number of tube
Nu	Nusselt number
NTU	number of transfer units
p	pressure, N/m ²
Q	heat transfer rate, W
Re	Reynolds number
S	pitch, m
S_v	surface to volume ratio, m^{-1}
T	temperature, K
U	overall heat transfer coefficient, $W/m^2 K$
V	volume, m^3
ε	heat exchanger effectiveness
ϕ	impact angle of stream lines, degree
μ	dynamic viscosity, kg m/s
ρ	density, kg/m^3
ψ	porosity

Subscripts

a	air
avg	average
d	dead

i	inlet conditions
min	minimum
max	maximum
n	normal
o	outlet conditions
p	parallel
s	surface
t	total
w	water

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